2002

# A Theoretical Comparison Of The Mechanical Control Behavior Of A R744- And A R134a Automotive AC Compressor

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C18-2

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**Abstract**

A theoretical study has been carried out in order to understand the difference in mechanical control characteristic of a R744- and a R134a automotive AC compressor. A R134a compressor starts to de-stroke when the crank house pressure is 1 to 2 bar higher than the suction pressure. The R744 starts to de-stroke when the crank house pressure is 10 to 12 bar higher than the suction pressure. This causes some additional precaution for the R744 compressor. The theoretical analysis was carried out at a comparable temperature operating condition.

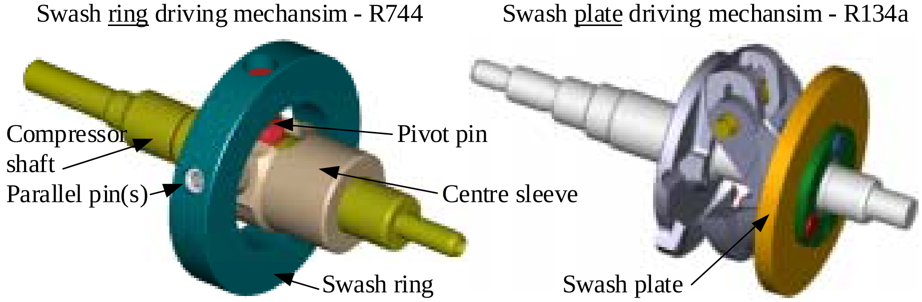
## Introduction

Higher pressure difference between the crank case pressure and the suction pressure is needed for a R744- compared to a R134a variable displacement compressor for automotive AC compressor in order to start the de-stroking process. It is desired to understand the reason for it and to find out if it is possible to reduce the pressure difference for the R744 compressor. A smaller pressure difference is to prefer since it cause less load on the piston bridge during the suction process and less strict requirements of the electronic control strategy.

## Compressor types included in the study

Two different driving mechanisms were investigated. They apply the same types of pistons and piston shoes but have different mechanism for varying the swept volume. The compressor type which already is widely in use - is the swash plate compressor. A new compressor type, which is ready for commercialisation is denoted swash ring compressor. The following figures show the two types of driving mechanism.

Swash ring driving mechansim - R744　　Swash plate driving mechansim - R134a

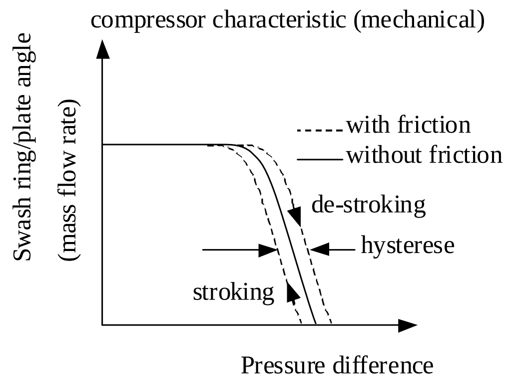



There are several advantages of a swash ring compressor compared to a swash plate. The swash ring driving mechanism is simple to assemble and to manufacture. The mechanical control hysterese i.e. pressure difference between the de-stroking- and the stroking process (see the following sketch of a control characteristic) is small based on low friction between the movable parts. The clearance volume can be kept constant independent of the swash ring angle. The rotating parts of the driving mechanism can easily be balanced out.

**Necessary crank case pressure to stroke- and to de-stroke the compressor**

The crank house pressure is applied to control the cooling / heating capacity of the compressor i.e. the mass flow rate. The mass flow rate is a measure of the cooling/heating capacity and is determined by the swept volume. The swept volume is determined by the swash ring/plate angle. The swash ring/plate angle will be applied here as a measurement of the compressor capacity. The compressor runs at maximum capacity when the crank house pressure is equal to the suction pressure. The compressor starts to de-stroke at a certain pressure difference i.e. between the crank case- and the suction pressure. The following figure shows schematically the swash ring/plate angle (synonym to mass flow rate) versus pressure difference (between the crank case and the suction pressure).

The figure shows a “good” mechanical compressor characteristic. The slope of the stroking and de-stroking is negative in the whole capacity range i.e. from maximum to minimum swash ring angle. The hysterese is small and the compressor starts to de-stroke with a small pressure difference.

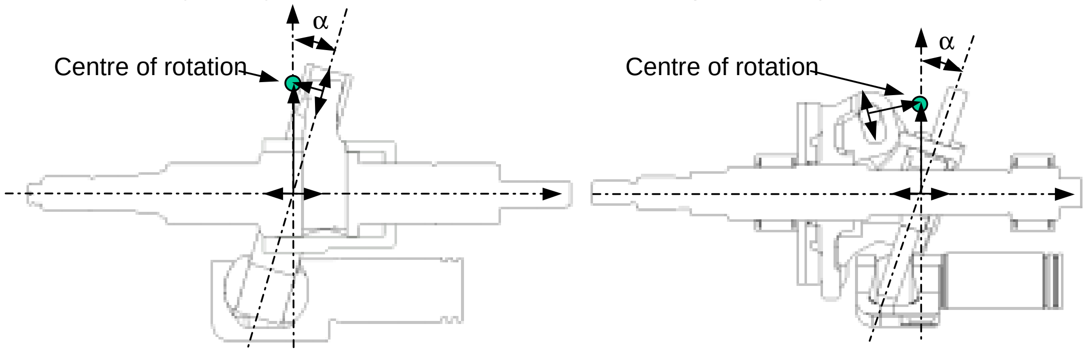



The horizontal part of the control characteristic (*the pressure difference increase without any change in the swash ring/plate angle*) is larger for a R744 compared to R134a. This demands some more attention for the electronic controller unit. Another aspect is the load increase on the piston bridge.

Several geometric parameters must be established in additional to an operating condition before such curve can be derived. The starting point is to establish the <u>swash ring/plate centre of rotation</u>, which is only geometric parameter dependent. The second step is to establish a <u>moment balance around the centre of rotation</u>. These two points will be worked out in the following sections.

**Swash ring/plate centre of rotation**

The swash ring/plate centre of rotation is necessary to establish in order to calculate the necessary crank case pressure, which is applied to de-stroke- or to stroke the compressor. This centre of rotation is not constant but dependent on the swash ring/plate angle (denoted $\alpha$). The centre of rotation is located where two normal (gliding) forces intersect. One force acts in the centre of the swash ring/plate (normal to the shaft axis). The other one is the normal force in the hinge mechanism, which is located eccentric to the compressor shaft axis. The following figures show schematically the centre of the rotation for a swash ring- and a swash plate compressor at maximum swash ring/plate angle. The vertical axis going through the shaft- and the swash ring/plate centre line is parallel displaced depending on the swash ring/plate angle.




The length from the shaft centre line to the centre of rotation becomes smaller with a decreasing swash ring angle.

**Moment balance around the centre of rotation**

A several forces are acting on the swash ring/plate, which create moments around the swash ring/plate centre of rotation. The main contributors are:

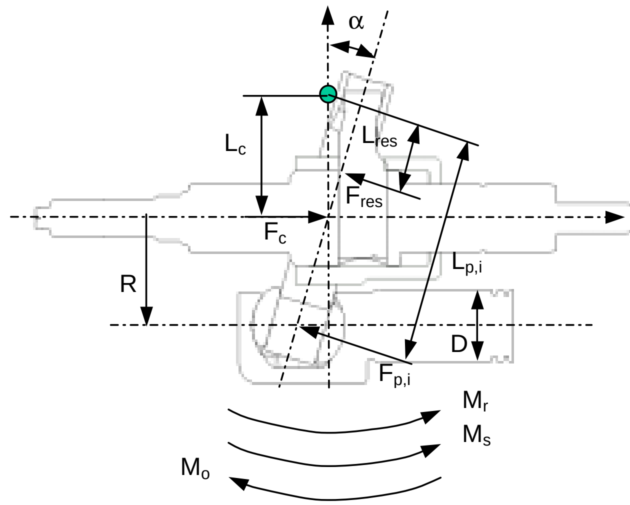
1. Resultant cylinder gas force and it's location ($M_{cylinder\ pressure}$): The cylinder gas force on each piston is summed up to a resultant gas force e.g. at an shaft angel interval of 1°. The location of the resultant gas force is established by a cylinder gas force moment balance around two axis (90° between each other), which are perpendicular to the compressor shaft angle. The resultant cylinder gas force tries to increase the swash ring/plate angle.
2. Resultant crank case gas force and it's location ($M_{crank\ pressure}$): The crank case gas pressure is the parameter of interest to establish. This gas pressure is given by the moment balance around the centre of rotation. The location if crank case gas force is in the centre of the compressor shaft. The resultant crank case gas force tries to reduce the swash ring/plate angle.
3. Moment of rotating masses ($M_r$): Compressor parts which are rotating cause a moment around the centre of rotation. This moment tries to reduce the swash ring/plate angle.
4. Moment of oscillating masses ($M_o$): Compressor parts which are oscillating cause also a moment around the centre of rotation. This moment tries to increase the swash ring/plate angle.
5. Reset spring force ($M_s$): This force acts in the same direction as the crank case pressure and tries to reduce the swash ring/plate angle.

A moment balance around the centre of rotation must be carried out to in order establish the magnitude of the crank house pressure, which gives the sum of the moments equal to zero. The moment balance around the centre of rotation can be written as shown bellow:

$$M_{crank\ pressure} = M_{cylinder\ pressure} + M_{oscillating\ mass} + M_{rotating\ mass} + M_{reset\ spring}$$

The following sketch shows the swash ring driving mechanism with the forces and their locations plus the moments acting on the swash ring.

The moment caused by the crank house pressure and the piston pressure can be written as:




$$M_{crank\ pressure} = I\ A_p\ P_c\ L_c = F_c\ L_c$$

$$M_{cylinder\ pressure} = A_p \sum_{i=1}^{I} P_{p,i}\ L_{p,i}$$

$$= F_{res}\ L_{res}$$

where:

| | |
|---|---|
| I | total number of pistons |
| $A_p$ | piston cross area |
| $F_{res}$ | resultant cylinder gas force |

**Operating condition and compressor parameters**

There are many combinations of temperatures, pressures, shaft velocity, air velocities etc. which define an operating condition. Different norms and standards on testing automotive ac system describes wind tunnel air velocity, humidity, air temperature etc. and not the system pressures for a specific ambient air temperature condition. One operating condition for a R134a system and a comparable operating condition for R744 is shown in the table to the right.

| **Parameter** | **R134a** | **R744** | **Units** |
|---|---|---|---|
| Evaporating temperature: | 5 | 5 | °C |
| Evaporating pressure: | 3.5 | 40 | bar |
| Compressor inlet temperature: | 15 | 25 | °C |
| High-side pressure: | 15 | 100 | bar |
| Condensing temperature: | 55 | | °C |
| Shaft velocity: | 1000 | 1000 | rpm |
| Pressure ratio: | 4.3 | 2.5 | - |
| Pressure difference: | 11.5 | 60 | bar |

*Ambient temp. was assumed to 45°C*

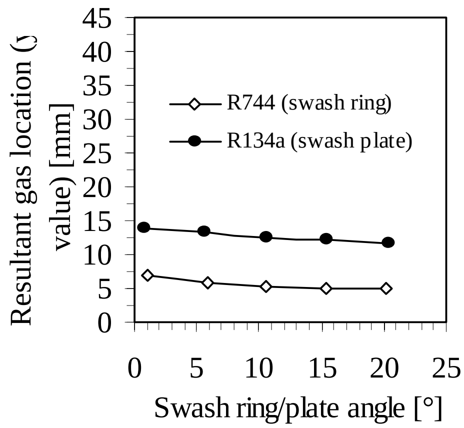
| **Parameter** | | **R134a** | **R744** | **Units** |
|---|---|---|---|---|
| Total swept volume ($V_{s,t}$): | | 160 | 33 | [cc] |
| Number of pistons (N): | | 7 | 5 | [-] |
| Piston diameter ($D_c$): | | 31.6 | 20 | [mm] |
| Max. swash ring/plate angle (α): | | 20.3 | 20.2 | [°] |
| Radius shaft- to cylinder centre line (R): | | 39.75 | 28.5 | [mm] |
| Hinge data: | hinge slot angle (γ): | 65.0 | - | [°] |
| | hinge pin location ($Y_s$): | 28,1 | - | [mm] |
| | hinge pin location ($Z_s$): | 24.8 | - | [mm] |

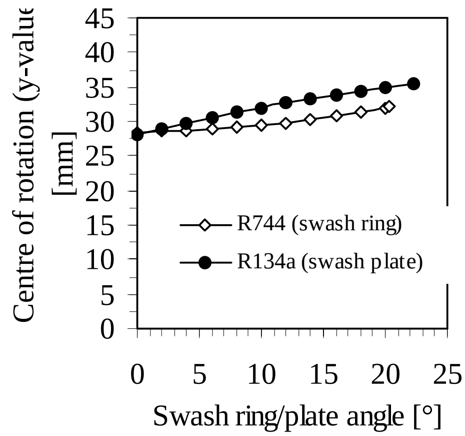
The compressor data for a R134a and a R744 is also shown to the right. The masses of rotating- and oscillating parts are not included since the influence of mass moments are relative small at low shaft speeds.

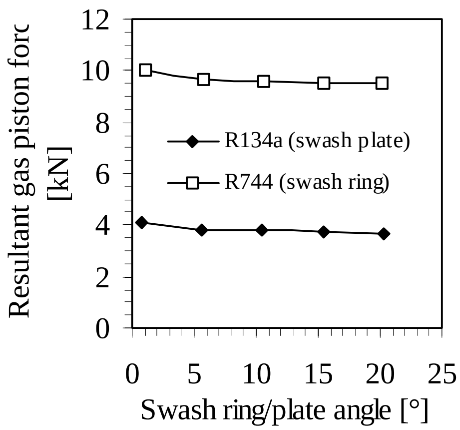
The influence of friction should be divided into two "groups". The first group is mechanical friction between parts, which are rotating and oscillating during steady operating condition (constant swash ring/plate angle). The second group is mechanical friction when the swash ring/plate angle either increase or decrease from a steady condition. However, friction will not be included in the calculations.

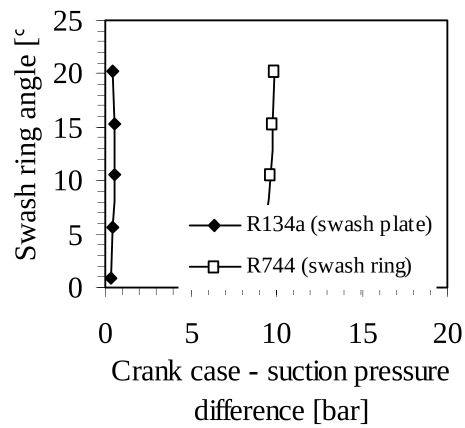
**Analysis of the mechanical control characteristic**

Losses in the compressors are not included such as friction, pressure drops, gas leakage and heat transfer. Forces caused be oscillating and rotating masses and the influence of the reset spring are also not included. The only forces acting on the swash ring/plate in this analysis are the piston gas forces and the balancing force caused by the crank case pressure. The moment balance around the centre of rotation is therefore: $M_{\text{crank pressure}} = M_{\text{piston pressure}}$. The calculation results based on the already mentioned operating condition and compressor data are presented in the following figures.




The calculations are in agreement with experimental results i.e. a R744 compressor needs considerably higher crank case pressure in order to start the de-stroke process compared to a R134a compressor. What is quite pronounced is the much higher resultant gas force from the pistons for the R744 compressor i.e. more than 2 times higher. The location of the resultant force (y-value) is on the other hand 2 times smaller. It must be emphasised that the curves of swash ring/plate angle versus pressure difference is not satisfying because of the slope (positive).

However, this is a result of neglecting the compressor and only taking the cylinder- and the crank house pressure into consideration.

**Analysis of the higher crank case pressure for the R744 compressor**

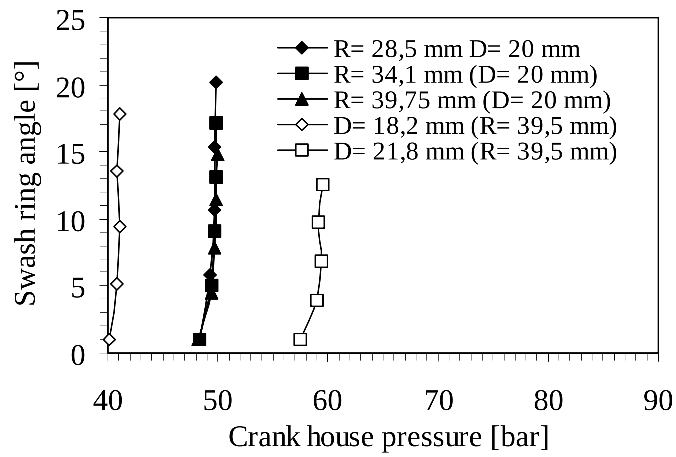
The only geometric parameter which has an influence on the crank case pressure are: piston diameter (D), radius from shaft- to cylinder centre line (R) and the swash ring angle ($\alpha$), see figure above. This is valid for:

1. a mass less compressor
2. an ideal compressor (no losses)
3 a set operating condition
4 a specific total swept volume ($V_t$= 0,5 $\pi$ I $D^2$ R tan $\alpha_{max}$)

The $\alpha$ is treated as a dependent variable, which is decided by the R and D. The values of "R" and "D" in the analysis are chosen so they are resulting in a maximum swash ring angle of about 20°. The results from the R and D parameter analysis are shown in the following table.

The maximum swash ring angle decrease when the radius (R) increase (constant cylinder diameter). The maximum swash ring angle ($\alpha$) decreases also when the piston diameter (D) increase (constant radius).

The figure shows that the radius between the compressor shaft- and the cylinder centre line (R) has no influence on the magnitude of the crank house pressure.




The piston diameter (D) has on the other hand a significant influence. Necessary crank house pressure in order to start the de-stroking process of the compressor increase with increasing piston diameter. The piston diameter determine the magnitude of the resultant gas force transferred to the swash ring.

The only moments acting on the swash ring in the analysis above is the cylinder gas pressure and the crank house pressure, which results in the following moment balance:

$$M_{\text{crank pressure}} = M_{\text{piston pressure}}.$$

This can also be written: $F_c L_c = F_{res} L_{res}$ (shown in an earlier section)

The analysis of the swash ring compressor shows that the ratio "$L_{res}/L_c$" is constant when the piston diameter is kept constant and by only altering the radius between the shaft- and the cylinder centre line. The resultant gas force on the piston changes with the piston diameter squared and the force is therefore sensitive in a change in the diameter.

The choice of the piston diameter must be made so the crank case pressure necessary to change the swash ring angle from maximum to minimum is always higher than the suction pressure.

**Other implication of the parameter "R", "α" and "D" on the driving mechanism**

The piston diameter (parameter "D") has a large influence on the forces acting on the compressor parts, which are rotating and oscillating. The side forces on the piston, the force transferred by the swash ring shoe to the swash ring, the main thrust bearing load, hysterese between de-stroking and stroking get larger with increasing piston diameter. There is an upper limit for the piston diameter (or with other word the piston gas force) in order to achieve the desired lifetime of the compressor without resulting in a too large overall physical size of the compressor.

The radius between the cylinder- and the shaft centre line (parameter "R") has not such a significant influence on the driving mechanism as the piston diameter. The bending of the shaft and the bending of the pivot pin get larger with increasing radius. This is valid when the design of the shaft and the pivot pin is kept unchanged. The radius has on the other hand a larger effect on the physical size of the compressor i.e. the outer compressor diameter.

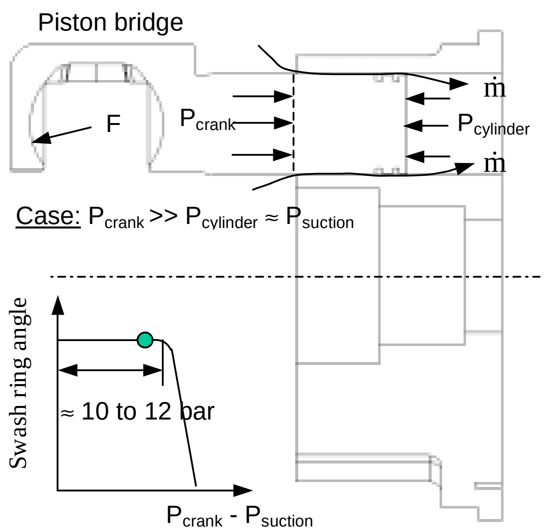
The maximum swash ring angle (parameter "α") has an influence on the overall compressor size and the driving mechanism. The side load on the piston get larger with increasing maximum swash ring angle i.e. same effect as making the piston diameter larger. There is also an upper limit for the maximum swash ring angle, which is determined by the joint (combination) of the swash ring shoe and the piston shoe recess design. The maximum swash ring angle has also an influence on the physical size of the compressor i.e. the outer compressor length.

**Implications of a "large" pressure difference between suction and crank case pressure**

There are several aspect which must be investigated when a "large" pressure difference is needed to start the de-stroking process e.g. the control strategy, piston bridge strength, gas leakage etc. The figure to the right shows:

# a cut-away drawing of one piston and a cylinder block
# a principle sketch of a mechanical compressor control characteristic

The control strategy should be made preferably so that the compressor is running with a pressure difference close to the pressure difference needed to start to de-stroke the compressor. This would decrease the response time for a fast change in capacity. Such a point is marked as a round circle



in the control characteristic sketch. This feature behaviour is solved without any major effort by the system supplier. The control strategy is based on thoroughly theoretical study, which is verified with experimental tests. The tests have revealed stable behaviour and fast response on a desired change in capacity.

The piston bridge is exposed for a larger load (denoted by parameter “F” in the last figure) during the suction process when the pressure difference is large. This demands a more careful design of the bridge in order to keep the deflection and the bending on an acceptable level so the desired lifetime is reached. Compressor simulation programmes including the influence of masses and compressor losses are applied in order to establish the load on the piston for different operating condition. The load results are then applied as input data to Finite Element Method (FEM) programme in order to establish stress and deflection. Duration tests have been carried out at different operating condition without any occurrence of piston bridge problem

The gas leakage from the crank case into the cylinder volume via the pistons should in theory be higher at a larger pressure difference. A R744 compressor apply piston (sealing) rings, which is designed to seal “good” against 110 bar gas pressure difference. A pressure difference of 10 to 15 bar between the crank case and the cylinder is therefore not of any problem. Experimental tests have also shown practically no difference in volumetric efficiency when a compressor is running with zero pressure difference compared to a pressure difference of 10 bar.

**Summary**

Necessary pressure difference between the suction- and the crank case pressure to start the de-stroking process of R744 based on experimental tests is in the range of 10 bar (depending on the operating condition). This pressure difference is considerable larger compared to a R134a compressor. The theoretical analysis was focused on the swash ring concept (R744) and compared with results from a R134a compressor.

Necessary crank house pressure to de-stroke the compressor is determined by the swash ring angle, the piston diameter and the radius between shaft- and cylinder centre line (for a specific operating condition and total swept volume). The swash ring angle was treated as a dependent variable, which leaves the radius and the diameter as free variables. The theoretical study of the swash ring concept revealed that the radius parameter has no influence on the crank house pressure. The piston diameter has a significant influence on the necessary crank house pressure to start the de-stroke process. The piston diameter determine the magnitude of the resultant gas force from the pistons acting on the swash ring. The resultant cylinder gas force becomes higher with increasing diameter, which results in a higher crank case force in order to balance out the cylinder gas force.

All three parameters have an upper and a lower limit given by desired physical compressor size and maximum allowable forces in order to reach desired compressor lifetime. A compromise has been found for the R744 compressor based on physical compressor size, compressor controllability and lifetime of the compressor parts.